

Development of a Forced Roll Mechanism for Planing Hull Models

ABSTRACT

A Forced Roll Mechanism was designed and built at the Hydromechanics Laboratory at the United States Naval Academy. The FRM was designed as a dynamometer that forces a planing hull model in roll and measures the resulting roll moment as well as the heave and sway forces. The model is fully restrained in all six degrees of freedom. The dynamometer can be configured to either oscillate the model or hold the model at a fixed roll angle, and measure the forces whether dynamic or static in nature, while allowing different testing conditions in terms of speed, rise and trim. The FRM was bench tested using known forces and moments to ensure accuracy and the measured lift forces for zero roll tests were compared with analytical predictions. Initial testing results show that the added inertia hydrodynamic coefficient depends on roll oscillation frequencies for low frequencies. The rig construction and calibration as well as preliminary results are presented.

INTRODUCTION

The operational speed of high-speed planing boats can be limited by the ability of the personnel to withstand the shock and vibration caused by slamming in rough water at high speed. High-speed planing boat operators are subject to periodic vibration and repeated impact loads that are of sufficient magnitude to cause fatigue, discomfort, and occasional injury. Therefore, removing the operator would allow high-speed planing boats to operate at higher speeds. However, dynamic instabilities, such as porpoising, chine-walking, steady heeling, and bow diving, are often speed dependent and are

more common at higher speeds. These instabilities are currently corrected for by the operator based on training and personal experience. Without an experienced operator controlling the boat, the operational speed would be limited by these dynamic instabilities. A better understanding of the dynamics and hydrodynamics involved with these instabilities would allow better predictions and control methods for avoiding these behaviors while operating at high-speeds for unmanned vessels.

The problem of dynamic stability of high speed planing craft has been known for many years. In the early 1930's, von Karman (1929) and Wagner (1931) performed analytical studies of planing hydrodynamics with respect to seaplane landings. Codega and Lewis (1987) described a class of high-speed planing boats that exhibited dynamic instabilities such as the craft trimming by the bow, rolling to a large angle of heel to port, and broaching violently to starboard. Blount and Codega (1992) presented data on boats that exhibited non-oscillatory dynamic instabilities and suggested quantitative criteria for development of dynamically stable planing boats. However, the mechanism of these instabilities is still not understood. Additional research has been done into steady state, calm water planing craft performance, yet not much attention has been given to dynamic stability and seakeeping (planing dynamics). This is due to the difficulty involved in determining the time dependent forces acting on the hull. To solve the equations of motion for a planing hull, accurate descriptions of the forces acting on the hull are needed. Some previous work has investigated planing seakeeping and dynamic stability in the vertical plane (porpoising behavior). Martin

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(1978a, 1978b) used a strip-theory based on slender body approximations to develop coupled linear heave/pitch/surge equations of motion. Troesch and Falzarano (1993) studied the development of unstable behavior in the vertical plane (heave and pitch) using both linear and nonlinear analysis methods. However, little work has been done on transverse dynamic stability.

An experimental program at the United States Naval Academy has been designed to determine the transverse plane stability of planing hulls. An experimental mechanism to force a planing hull model in roll motion was designed and built. This paper presents a description of the design, construction and calibration of the Forced Roll Mechanism, as well as some preliminary results. A photographic overview of the Forced Roll Mechanism (FRM) is shown in Figure 1. Bench testing of the FRM consisted of using calibrated weights to produce known forces and moments allowing evaluation of the mechanism's accuracy.

Once the bench testing was completed, the Forced Roll Mechanism was attached to the towing carriage. Initial testing in the tank was done with a 20° deadrise prismatic planing hull model fixed in pitch, heave, roll and sway and towed through a range of constant speeds in calm water. The resulting lift forces were compared with the predictions of lift from Savitsky (1964) and Savitsky and Brown (1976). This established high confidence that the FRM was accurately measuring vertical forces. These tests were followed by a series of static tests with the model fixed in roll and dynamic tests with the model oscillated in pure roll. The static roll tests were used to determine the roll stiffness and the dynamic tests were used to determine the added inertia and damping due to hydrodynamic forces on the tested prismatic form.

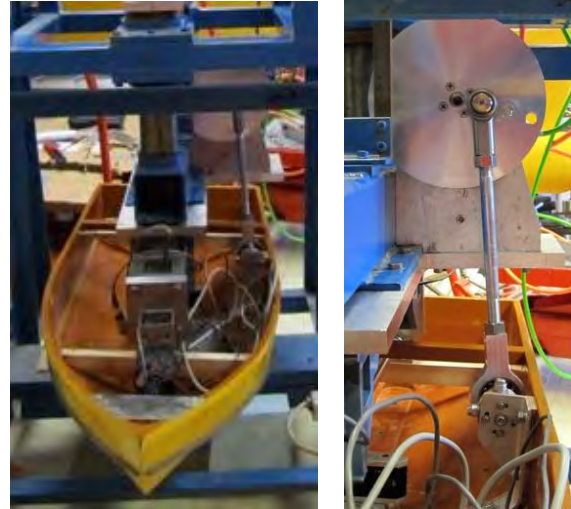


FIGURE 1: Overview of Forced Roll Mechanism and Detail of the Roll Forcing Design

DESIGN DESCRIPTION

The FRM is intended to act as a dynamometer that forces a planing hull in roll and measures the resulting roll moment as well as the heave and sway forces. The trim and rise of the model can be adjusted between test runs. The dynamometer can be configured to either oscillate the model or hold the model at a fixed roll angle, and measure the forces whether dynamic or static in nature, while allowing different testing conditions in terms of speed, rise and trim.

The FRM mechanism was designed to oscillate the model up to 30° at a frequency of 3 Hz. The maximum expected roll moment for this case was 250 in-lbs. At the higher model speeds and oscillation frequencies, the forces exceeded the force capabilities for the bi-axial load cell. Therefore, the maximum oscillation frequency was 2.5 Hz. Since the model used for initial testing of the FRM has a deadrise of 20°, the maximum roll amplitude used was 25°.

The dynamic roll system consists of a Baldor servo-motor with a planetary gear box attached to a ¾ inch aluminum circular plate. A picture of the roll forcing design is shown in Figure 1. A connecting arm made from a 5/8 inch aluminum turnbuckle connects the circular plate to the

model at the gunnel on the port side of the model. The connecting arm ends in a self-aligning ball bearing connected to a bi-axial load cell. The bi-axial load cell is oriented to record vertical and horizontal (side) forces relative to the hull. The connecting arm can be located at discrete distances from the center of the circular plate, allowing for varying amplitudes of roll motion. The center of rotation for the model is a 5/8 inch precision ground aluminum rod running parallel to the keel. The rod is fixed with respect to the model and is connected to the supporting system with pillow block needle bearings. The roll position feedback is determined by an Angular Displacement Transducer (ADT) fixed to one end of the rod by a bellows coupling. This allows a change in pitch of the model of up to 7° relative to the stationary ADT. The heave and sway forces are recorded through the bi-axial load cell and by two sets of two four inch block force gauges. The block force gauges are made by Hydronautics, Inc. and are mounted to a Heave Post Assembly also built by Hydronautics, Inc. The heave is controlled through positioning the FRM at the desired vertical position. The trim of the model is controlled through adjusting bolts in slots on the plate attached to the aft force gauges.

With the present motors and instrumentation, the system is capable of a maximum RPM of 180 (3 Hz) and a maximum roll amplitude of 30 degrees for the 20° deadrise wooden planing hull model. The roll angle amplitude is limited by the model clearance with the FRM support structure. The force gauges are rated to 50 lbs for sway forces and 100 lbs for vertical forces. Judge (2010) showed that the roll added inertia calculation is highly sensitive to the error in the measured forcing moment amplitude at low frequency. Therefore, to accurately determine the coefficients in roll when oscillating at low frequencies, there must be tighter error control of the forcing moment measurements or an increased number of test runs. To address this

issue, the force gauges can be replaced with gauges of greater sensitivity when doing testing at low roll frequencies.

By looking at groups of load cells, the heave and lift forces and the roll moment can be found. The block force gauges attached to the supporting heave post assembly measure purely vertical and horizontal (sway) forces in the tow tank reference frame. However, the bi-axial load cell is attached to the model and, therefore, reads vertical and horizontal (side) forces relative to the model. Since the model is pitched and rotating in roll, these measured forces need to be resolved to find the vertical and horizontal forces relative to the heave post assembly.

FORCES AND MOMENT CALCULATIONS

Figure 2 shows a schematic of the measured forces for an upright model as well as a model at a non-zero roll angle and fixed trim angle.

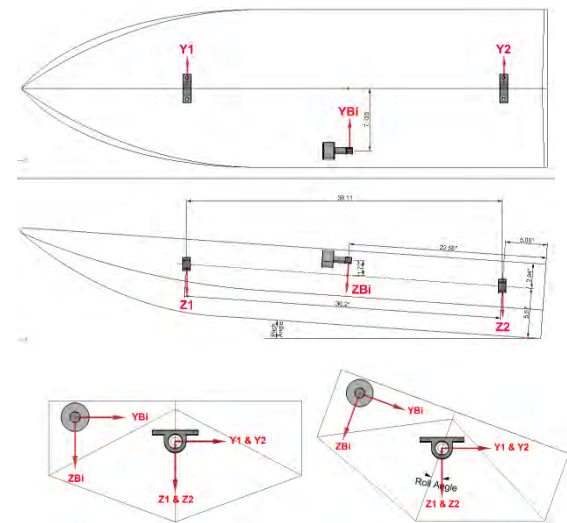


FIGURE 2: Schematic of Force Measurements

The following quantities are defined as:

- Y_1 horizontal load cell measurement at forward block gauge
- Y_2 horizontal load cell measurement at aft block gauge

- Y_{Bi} bi-axial load cell measurement parallel to model “deck”
 Z_1 vertical load cell measurement at forward block gauge
 Z_2 vertical load cell measurement at aft block gauge
 Z_{Bi} bi-axial load cell measurement perpendicular to model “deck”

The forces measured from the bi-axial load cell must be resolved into vertical and horizontal forces relative to the block forces gauges. For a given roll angle, ϕ , and trim angle, τ , the individual forces from the bi-axial gauge are resolved into a total force at a particular direction (relative to the deck edge of the model). This resulting force is in a plane intersecting the model that is parallel to the transom (and, therefore, at an angle τ relative to vertical). The resulting force in this plane is

$$|F_\tau| = \sqrt{Y_{Bi}^2 + Z_{Bi}^2} \quad (1)$$

at angle $\gamma = \tan^{-1} \frac{Y_{Bi}}{Z_{Bi}}$ relative to the Z_{Bi} force direction. For a given roll angle, ϕ , the angle relative to vertical within this plane is

$$\xi = \gamma + \phi. \quad (2)$$

Therefore, within this plane, tilted at the trim angle, τ , the vertical and side forces are

$$F_{\tau z} = |F_\tau| \cos \xi \quad (3)$$

$$F_{\tau y} = |F_\tau| \sin \xi \quad (4)$$

respectively. Finally, resolving these forces into the vertical and sway forces in the earth reference frame gives

$$Y_3 = F_{\tau y} = |F_\tau| \sin(\gamma + \phi) \quad (5)$$

$$Z_3 = F_{\tau z} \cos \tau = |F_\tau| \cos(\gamma + \phi) \cos \tau. \quad (6)$$

To find the total heave force, the vertical forces from all the force gauges are summed,

$$Total F_{heave} = Z_1 + Z_2 + Z_3. \quad (7)$$

To find the total sway force, the horizontal (side) forces from all the forces gauges are summed,

$$Total F_{sway} = Y_1 + Y_2 + Y_3. \quad (8)$$

To find the roll moment, the resulting roll moments due to the forces acting at the bi-axial load arm connected to the model need to be summed,

$$M_{roll} = Y_{Bi} \cdot d_y - Z_{Bi} \cdot d_z \quad (9)$$

where d_y and d_z are the horizontal and vertical distances from the forces measured by the bi-axial load cell to the rotation rod (1.72 inches and 7.125 inches, respectively).

INITIAL TESTING

The first model tested was a wooden prismatic planing hull with a constant deadrise of 20° , a beam of 1.47 ft (0.45 m), and a total length of 5 ft (1.52 m). The model is marked in one inch increments at the chine with every fifth mark labeled. The model was mounted to the FRM and the initial testing was done first in air and then in water.

The objectives of the initial static testing of the dynamometer were to evaluate the mechanism's ability to measure forces and moments, assess the ability to reproduce the well documented lift results of Savitsky (1964) and Savitsky and Brown (1976), and to experimentally determine the amplitude and frequency range of forced oscillations for dynamic testing.

The testing for the first objective was done using calibrated weights. The known forces and moments produced by the weights were then compared to the measured forces and resulting moments from the FRM. This static testing showed the ability of the dynamometer to measure heave and sway forces as well as the roll moment at various trim and roll angles.

The following parameters for identifying system accuracy, as defined in Ashcroft et al. (1989), were used:

$$\frac{|F_{V_measured} - F_{V_actual}|}{F_{V_actual}} \quad (10)$$

$$\frac{|F_{H_measured} - F_{H_actual}|}{F_{H_actual}} \quad (11)$$

$$\frac{|RM_{measured} - RM_{actual}|}{RM_{actual}} \quad (12)$$

where F_V is the total vertical force, F_H is the total horizontal force, and RM is the total roll moment. The subscript *measured* refers to the measured force or moment and the subscript *actual* refers to the force or moment applied. The errors in measurement were small with errors less than 4%. The error measurements are shown in Figures 3 through 5.

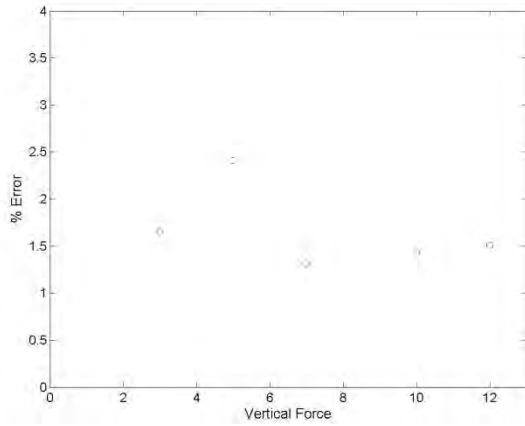


FIGURE 3: Error Measurements for FRM Vertical Force Measurements

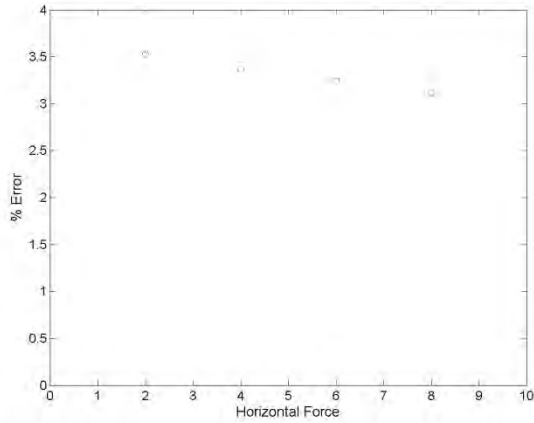


FIGURE 4: Error Measurements for FRM Horizontal Force Measurements

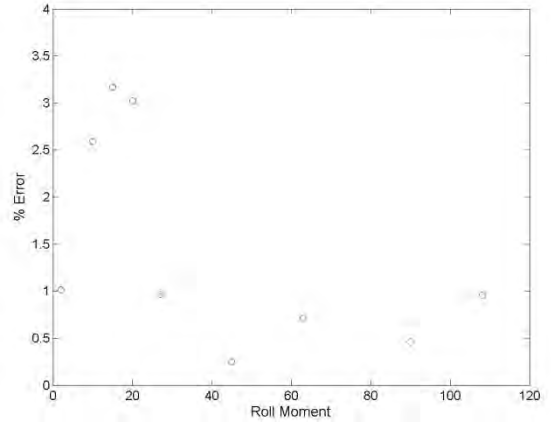


FIGURE 5: Error Measurements for FRM Roll Moment Measurements

The next phase of testing was done with the model in the towing tank. The results of the vertical lift force were within about 10% of the vertical lift predicted with the formulas of Savitsky (1964) and Savitsky and Brown (1976). The prediction equations from Savitsky (1964) and Savitsky and Brown (1976) are as follows:

$$\lambda = \frac{L_k + L_c}{2b} \quad (13)$$

$$C_v = \frac{V}{\sqrt{gB}} \quad (14)$$

$$C_{L0} = \tau^{1.1} \left[0.0120\lambda^{0.5} + \frac{0.0055\lambda^{\frac{5}{2}}}{C_v^2} \right] \quad (15)$$

$$C_{L\beta} = C_{L0} - 0.0065\beta C_{L0} \quad (16)$$

$$F_{vert} = C_{L\beta} \cdot \frac{\rho}{2} V^2 b^2 \quad (17)$$

The lengths for the wetted keel length (L_k) and for the wetted chine length (L_c) were taken from the measured wetted lengths during the test runs. The differences between the measured lift force and the predicted lift force increases as the trim angle decreases. This is because the prediction formulas assume a purely prismatic shape, i.e. the wetted surface has straight edges, while the actual model has curvature near the bow.

Therefore, the wetted surface edges on the actual model show curvature (see Figure 6).

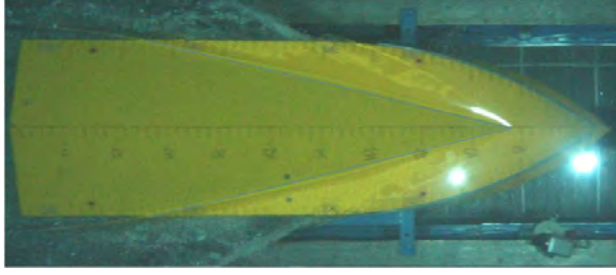


FIGURE 6: Photograph of Curvature of Wetted Surface Edges for Small Trim

The initial dynamic testing included oscillating the model in air to find the system roll moment of inertia. This system roll moment of inertia is needed so that it can be separated from the hydrodynamic added roll inertia. The first phase of the in-water test matrix consisted of towing the model at various fixed roll angles for two displacements, and three model speeds. The second phase of the in-water test matrix consisted of forcing the model in roll at three model speeds, three roll amplitudes, four frequencies of oscillation, and two displacements.

PRELIMINARY RESULTS

The added inertia, damping and restoring forces can be represented as coefficients times the roll acceleration, roll velocity and roll angle, respectively. This notation allows the equation of motion for a vessel in roll to be written as

$$RollMoment(t) = (I_{roll} + A_{roll})\ddot{\phi}(t) + B_{roll}\dot{\phi}(t) + C_{roll}\phi(t) \quad (18)$$

where the coefficients, A_{roll} , B_{roll} , and C_{roll} , are found from the forced motion experiments. The mass moment of inertia of the model in roll is represented by I_{roll} . Some preliminary results for the added inertia for two model speeds and roll oscillation frequencies are shown in Figure 7. Figure 8 shows the added inertia for two roll amplitudes versus roll oscillation frequencies. The model speeds are in the planing regime and the roll amplitudes are in the linear range. The coefficients have been normalized by the beam, water density, and gravity. The frequencies and

speeds are non-dimensionalized. The speeds are shown by the associated volumetric Froude number.

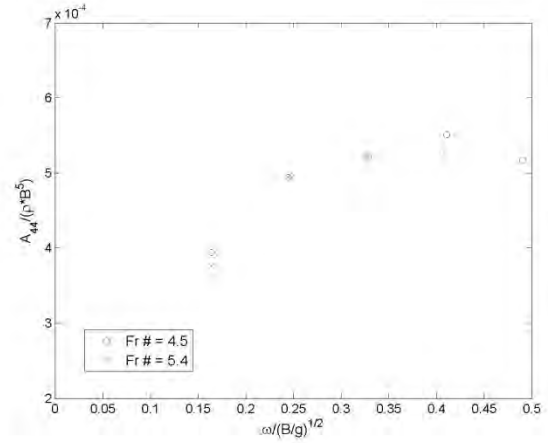


FIGURE 7: Added Inertia for 10° Roll Amplitude

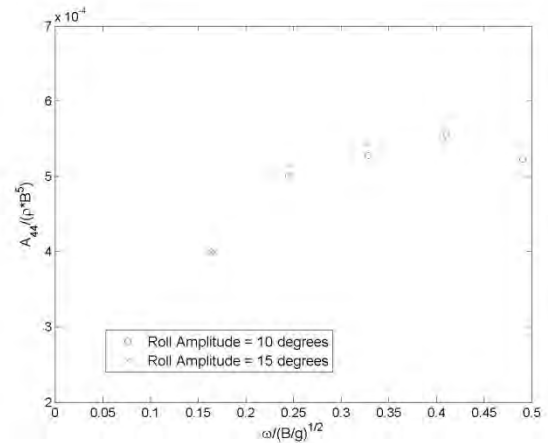


FIGURE 8: Added Inertia for Fr # = 4.5

The preliminary added inertia results show little dependence on model speed. However, there is a clear frequency dependence at lower frequencies. The added inertia in roll decreases at lower frequencies, yet appears less dependent at higher frequencies. In a study on transverse planing hull stability done at Steven's Institute of Technology (Brown and Klosinski, 1995), the authors state that since the support of a planing boat comes principally from dynamic pressure, it is largely independent of gravity effects and, therefore, the hydrodynamic added inertia for a rolling planing boat would be expected to be

independent of frequency. The Project Officer for the sponsoring agency did “not endorse the view that the added mass moment of inertia and the damping moment on a planing hull is independent of frequency.” Judge (2010) showed that at lower frequencies the added inertia calculation is more sensitive to measurement error. However, the percent standard deviation for the added inertia results at the lowest frequency were 3% or less, providing confidence in the frequency dependence of added inertia.

CONCLUSIONS

A Forced Roll Mechanism was constructed to allow a model to be forced in roll motion while being towed at planing speeds. The resulting heave and sway forces as well as the roll moment were measured and analyzed for dependency on roll amplitude, vessel speed, and roll oscillation frequency. Traditionally, theoretical planing analyses assumes the hydrodynamic coefficients of added inertia and damping are independent of frequency. However, initial test results showed that the added inertia coefficient, A_{roll} , depends on frequency for low frequencies of oscillation. This testing equipment will facilitate researching dynamic instabilities of planing craft in calm water and waves. The FRM provides a much needed and versatile platform for studying planing hull behavior in the transverse plane.

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